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DEVELOPMENT OF A HIGH-TEMPERATURE, NUCLEAR-RADIATION-RESISTANT PNEUMATIC POWER SYSTEM FOR FLIGHT VEHICLES

QUARTERLY REPORT

24 MARCH 1963

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DEVELOPMENT OF A HIGH-TEMPERATURE, NUCLEAR-RADIATION-RESISTANT PNEUMATIC POWER SYSTEM FOR FLIGHT VEHICLES

AF 33(616)-7582

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NOTICE

The information contained herein is advanced information and has not been approved by ASD. The USAF assumes no responsibility for the information or conclusions presented.

FOREWORD

This report, prepared by General Dynamics/Convair, covers research and development work accomplished under Air Force Contract AF 33(616)-7582 between 25 December 1962 and 24 March 1963.

USAF Contract AF 33(616)-7582 was initiated under Project Task No. 61085 by the Flight Accessories Laboratory, Aeronautical Systems Division, Wright Patterson Air Force Base, Ohio. The work is administered under the direction of Mr. B. P. Brooks, ASRMFP-3, Transmission Technology Section, Flight Vehicle Power Branch of the Flight Accessories Laboratory. It is conducted by Convair uuder the direction of Mr. R. W. Casebolt.

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1 | INTRODUCTION

The development of a high-temperature, nuclear-radiation-resistant pneumatic power system for flight vehicles was initiated by the Flight Accessories Laboratory of ASD to advance the state of the art of pneumatic systems.

Future flight vehicles will be subjected to extreme thermal environments because of heat generated by supersonic speeds and/or the propulsion systems. This results in new requirements for components and auxiliary systems. It is, therefore, a goal of this program to design, develop and test a typical pneumatic system capable of operating under wide temperature excursions while in the presence of nuclear radiation.

The work accomplished during the first portion of the program indicated that subsequent effort should be directed toward the development and testing of a closed, high-pressure pneumatic system for powering aerodynamic controls. A ram air turbine-driven compressor was selected as the most feasible pneumatic power source.

In addition to the compressor, other components currently under development include a rotary actuator and servo control valve, pressure regulator, relief valve, accumulator, filter, check valve, as well as miscellaneous hardware. Subsequent to evaluation, these components will be tested as parts of a pneumatic system, under a simulated high-temperature environment.

2 | SUMMARY

The experimental development of components for the high-temperature pneumatic system is continuing. Preliminary experimental development of the compressor (Phase I) was completed. Testing of the gas-lubricated journal bearing at speeds of 71,400 rpm (surface velocity of 7,030 inches/second) is covered in this report. Convair concludes that the original concept of a high speed centrifugal compressor, using gas bearings, is feasible.

In addition, one actuator and servo valve was completed and is currently undergoing preliminary testing. Three accumulators were welded successfully using the electron beam process. However, the development of a satisfactory pressure regulator has suffered another setback as a result of a ruptured bellows assembly.

3 | WORK ACCOMPLISHED

COMPRESSOR AND DRIVE UNIT

Phase I, feasibility study and preliminary experimental testing of the turbine driven air compressor, consists of four study areas:

1. Bypass Valve
2. Compressor
3. Air Bearing
4. Labyrinth Seals

The detail design of the bypass valve and the compressor development study was completed prior to this quarterly period. Preliminary testing of individual compressor stages resulted in efficiencies 5 to 10 percent greater than indicated by previous experience.

The air bearings and labyrinth air seals were developed concurrently during Phase I, which was brought to a satisfactory conclusion as a result of testing conducted during this reporting period. The work accomplished represents a significant contribution to the state of the art of high-speed, gas-lubricated bearings. Testing of the externally-pressurized step bearing reached speeds of 71,400 rpm under varying pressure ratio and supply pressure conditions.

It is interesting to note that the journal surface velocity of 7,030 inches/second (400 miles/hour) and the demonstrated film stiffness of 800,000 lb/inch are both substantially greater than that of any known gas lubricated bearing

currently under development. The air bearing tests demonstrated excellent agreement with the predicted threshold of half-frequency whirl. No indication of instability was observed.

The experimental testing accomplished was sufficient to conclude that the general concept was feasible. A revised design is proposed for the Phase II portion of the compressor program.

Air Bearing Research

A series of preliminary tests were performed on a one-inch diameter model of the proposed air bearing configuration. This was necessary to verify the validity of the calculation procedures and bearing stability criteria.

Figure 1 gives the results of a critical speed analysis of this simple rotor bearing system. Stability criteria were applied to the data of Figure 1 to obtain Figure 2. The calculated threshold of half-frequency whirl instability is shown as the shaded line. Operation to the left of this line was predicted to be stable; operation to the right was predicted to be unstable. The black dots represent points (determined by experiment) of instability.

A series of step bearings with variations in step shaft clearances (C_s), step height (H) and step length (W_s) were tested. (See Figure 3 for a typical step journal bearing configuration.)

Except for these variations, the shaft and bearings were essentially geometrically similar to the 1.875-inch compressor bearing. The data obtained in Figure 2 was the final result of this initial test series. The predicted and experimentally-observed points of instability were in good agreement at speeds up to 90,000 rpm.

Modifications were then made to the computer program for the 1.875-inch diameter rotor assembly and performance predictions were undertaken. The

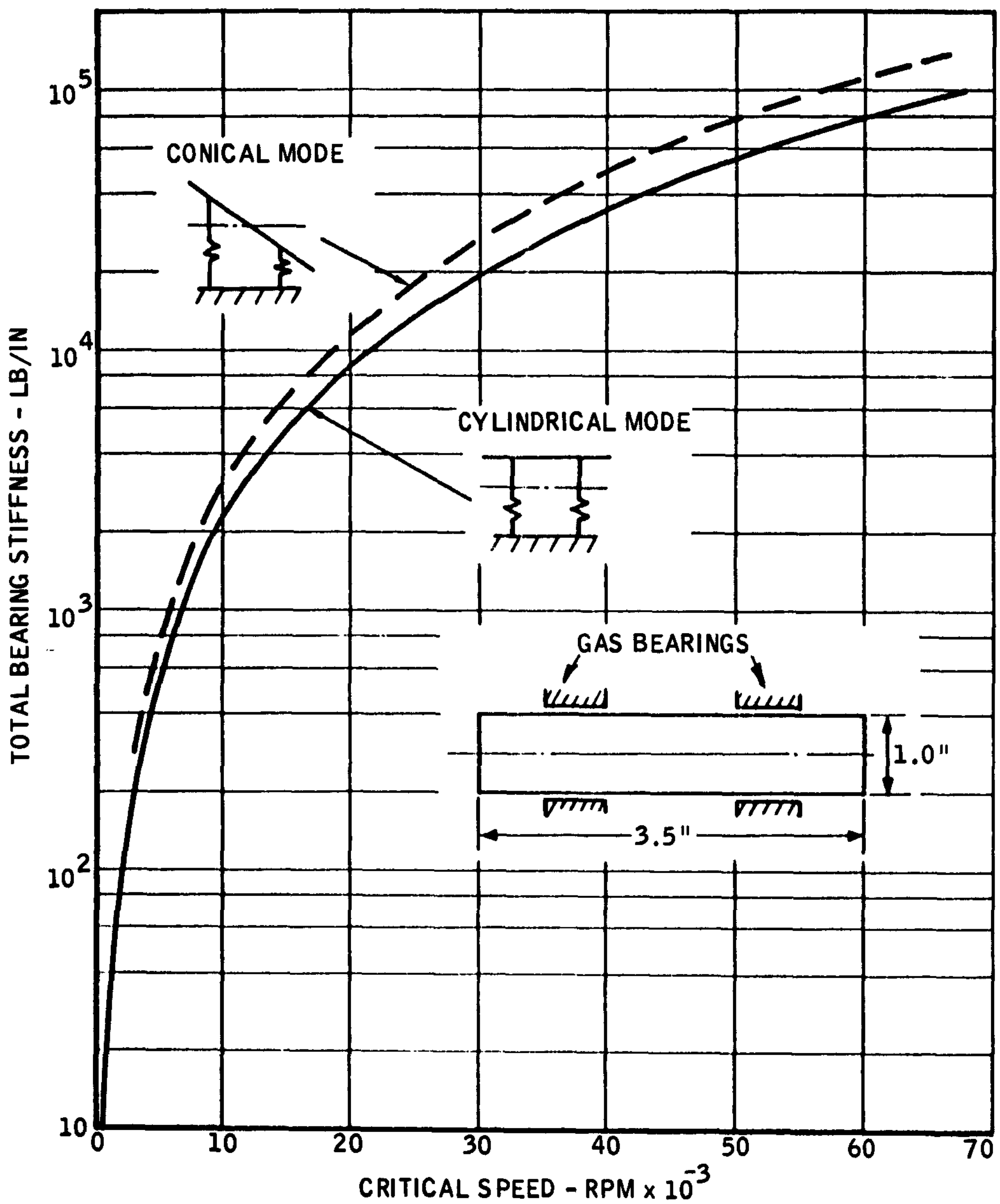

Figure 1. One-Inch Diameter Shaft - Bearing Stiffness vs. Critical Speed

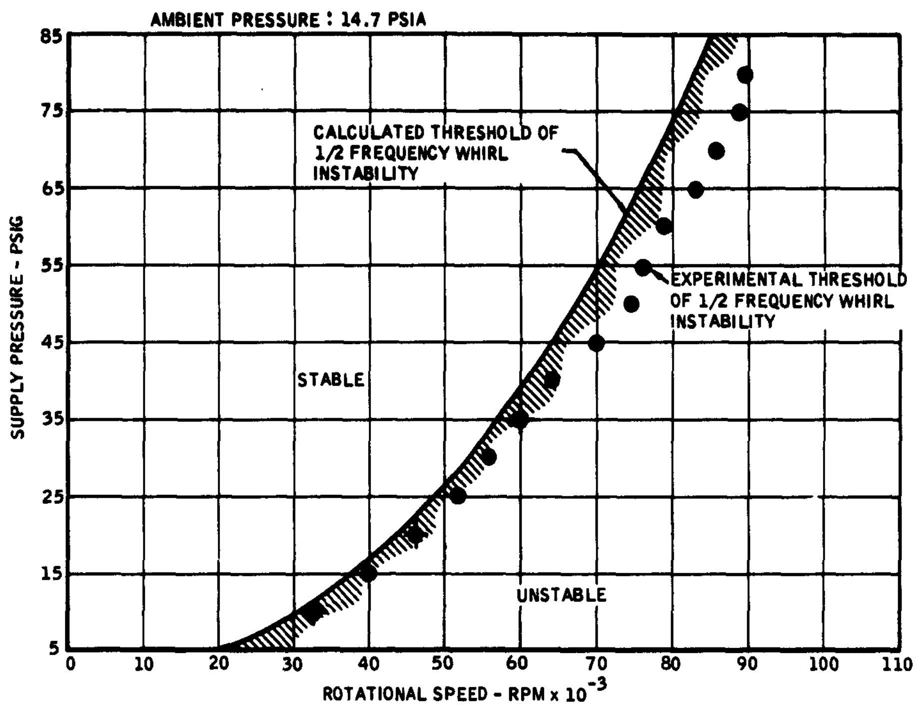

Figure 2. Stability Characteristics of a One-Inch Diameter Model Bearing

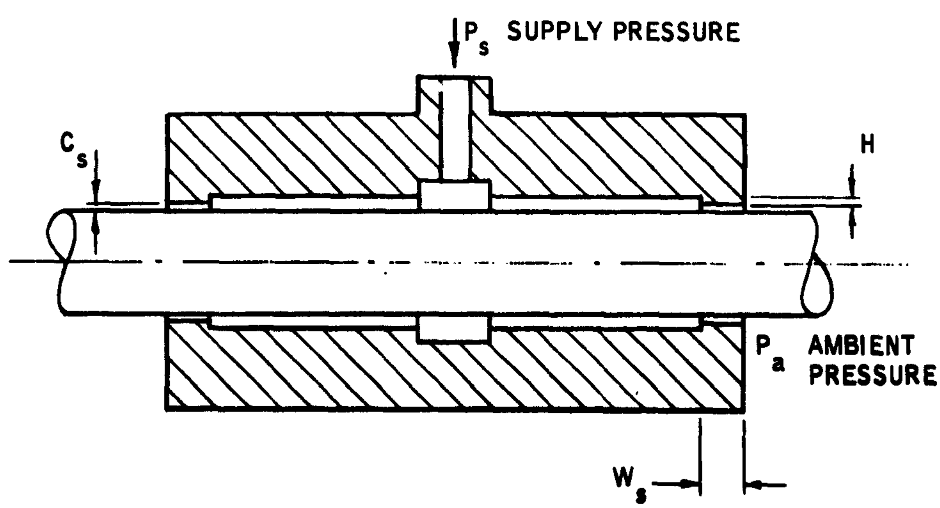

Figure 3. Externally Pressurized Step Journal Bearing

characteristics of a step bearing are very strongly dependent on the clearance; Figure 4 was, therefore, prepared to show the computed radial clearance as a function of speed. This information was then utilized in preparing a complete performance map of the rotor-bearing assembly as shown in Figure 5. A series of stable test points were taken at various pressure ratios, speeds, and supply pressures as plotted on Figure 5. Stable operation, with little or no sign of half-frequency whirl, was observed at each data point. The testing was halted at 71,400 rpm due to inadequate turbine power. Further investigation revealed that an extensive modification of the test turbine would be required to permit operation at higher speeds.

Typical air-bearing temperature is plotted as a function of rotational speed in Figure 6. The supply pressure was 1200 psia with ambient pressure at 400 psia. The temperature rise with increasing speeds indicates that this heating could become troublesome at the design speed. It is, therefore, anticipated that temperature rise may be of prime consideration for the experimental high-temperature air-bearing design (Phase II).

Sufficient knowledge was obtained during this preliminary experimental program to conclude that:

1. The general design concept of an ultra-high-speed dynamic compressor utilizing gas bearings is feasible.

2. The externally-pressurized, step-type gas bearing is recommended for use in this application. The stability of this bearing can be predicted with reasonable confidence.

3. The design layout, speed, and bearing dimensions of the turbine-driven compressor should be modified in the high-temperature air-bearing design.

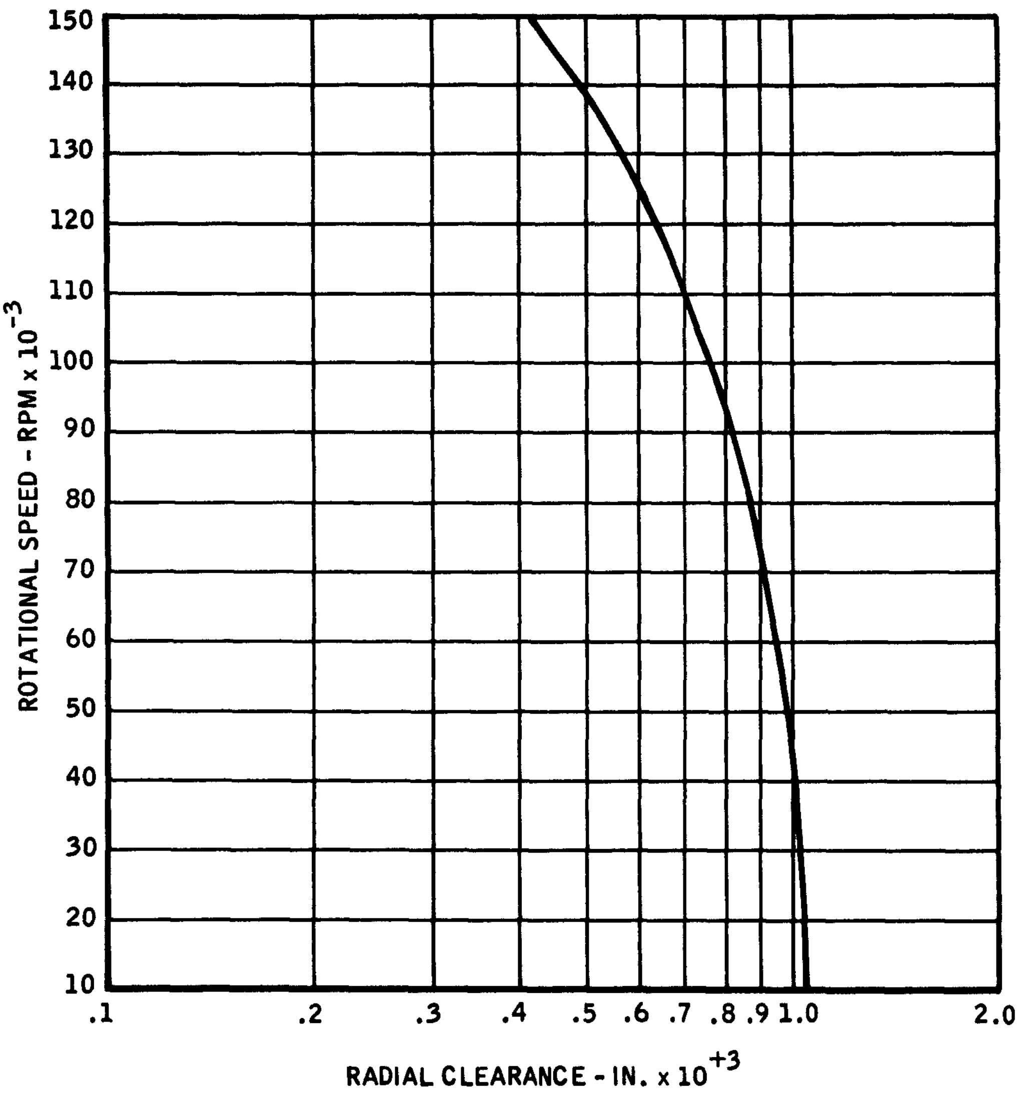

Figure 4. 1.875-Inch Diameter Air Bearing - Speed vs. Radial Clearance

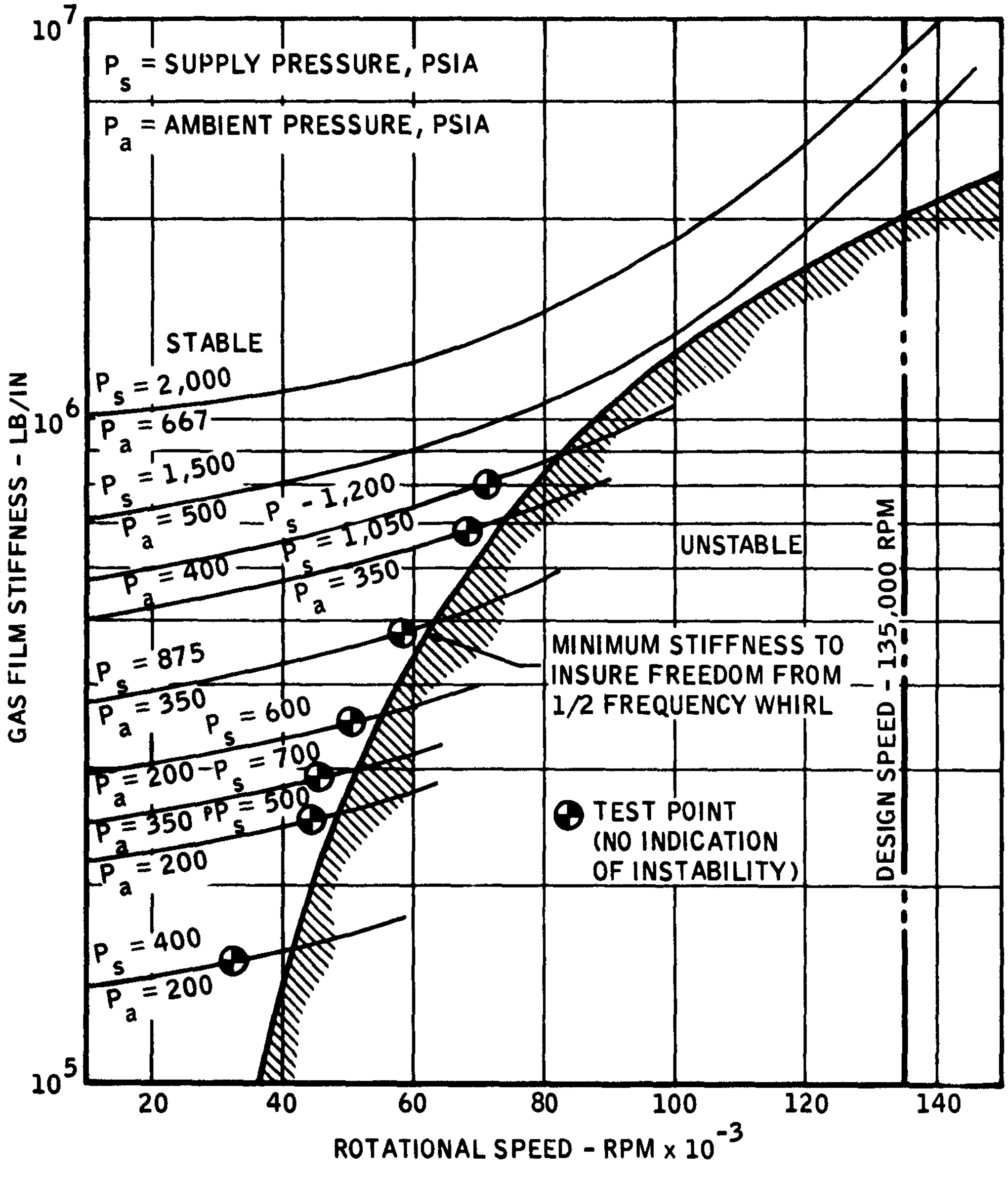

Figure 5. Stability Characteristics of 1.875-Inch Diameter Air Bearing

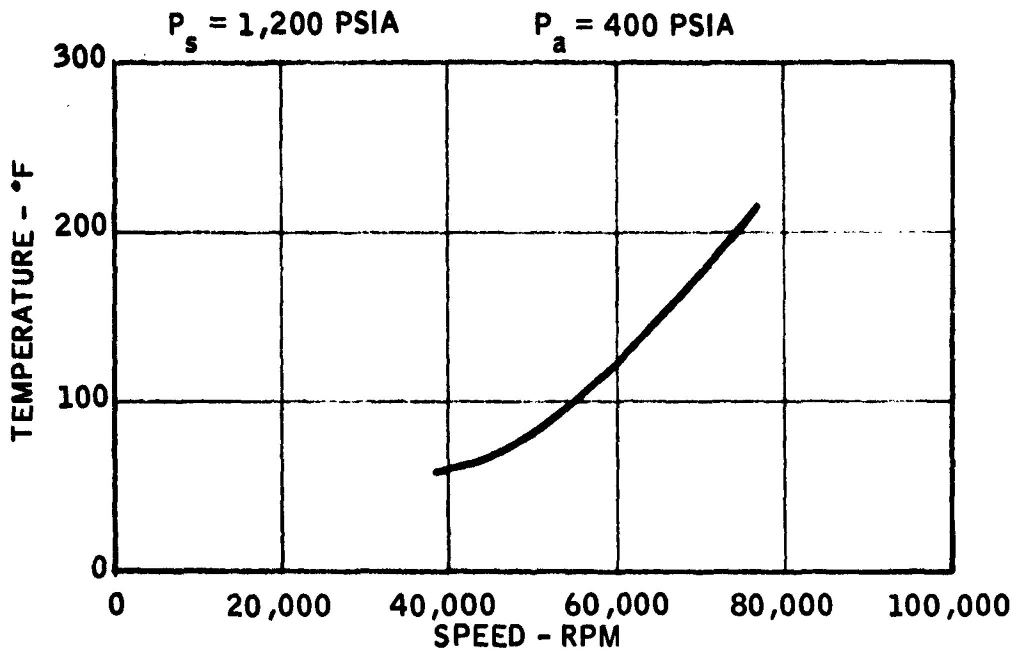

Figure 6. 1.875-Inch Air Bearing -- Temperature vs. Speed

Proposed Compressor Design (Phase II)

The performance of the experimental compressor, evaluated during Phase I, has resulted in sufficient data to establish the configuration of the final machine. The changes proposed for the Phase II compressor were made in order to reduce the problems concerned with the mechanical design. On this basis, the rotating groups were rearranged and the design speed was reduced from 125,000 rpm to 100,000 rpm. This can be accomplished without any penalty in overall performance of the unit.

The compressor characteristics of the proposed design are tabulated in Table I. As a result of the decreased speed, the first stage impeller diameter must be increased to 3.56 inches. Figure 7 is a schematic diagram of the Phase II unit. This general system arrangement changed very little from the preliminary arrangement. The basic difference is in the turbocompressor.

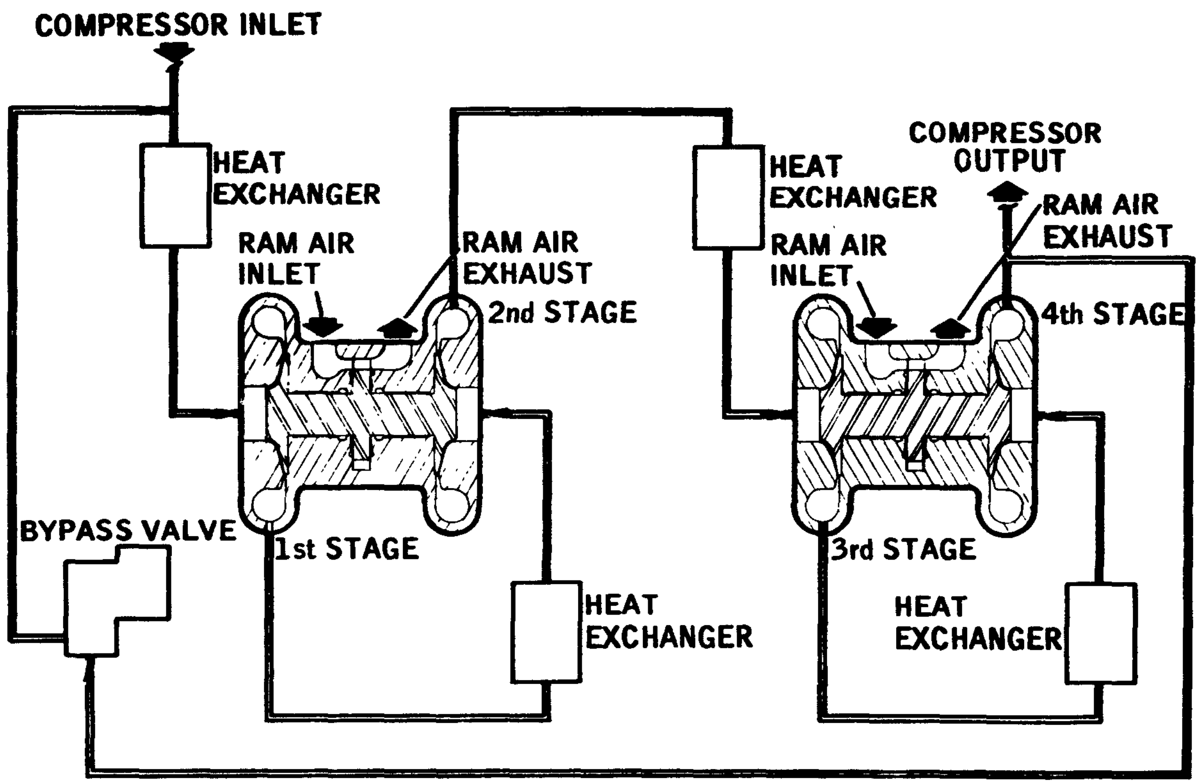

Figure 7. Phase II Turbocompressor — Schematic Diagram

TABLE I

COMPRESSOR DESIGN CHARACTERISTICS (PHASE II)

Required Pressure Ratio:	4:1	Inlet Pressure:	500 psia
Required Flow Rate:	20 lb/min	Outlet Pressure:	2000 psia
Speed:	100,000 rpm	Ambient Temp:	1000°F

STAGE NUMBER	1	2	3	4
Flow Rate, lb/min.	25	25	25	25
Inlet Temperature, °F	680	695	737	753
Inlet Pressure, psia	484	764	1122	1527
Pressure Ratio	1.60	1.48	1.37	1.31
Exit Pressure, psia	774	1132	1537	2000
Exit Temperature, °F	1000	1000	1000	1000
Stage Efficiency	0.48	0.42	0.40	0.37
Load Factor [a]	15.1	16.3	16.7	17.2
Impeller Diameter, in.	3.56	3.48	3.24	3.13
Tip Speed, ft/sec.	1555	1519	1413	1365
Required Power, hp	48.9	46.6	40.4	37.7

[a] Load Factor = $\dfrac{6.42\,(H_{ad})^{3/4}}{N\sqrt{Q}}$

where H_{ad} = Adiabatic head, ft.

N = Speed, rps

Q = Volumetric flow, cfs

The turbine drive is now centrally located between two externally pressurized journal bearings. This was done to reduce the stiffness required at the design speed (100,000 rpm).

Figure 8 shows the results of a critical speed analysis for this rotor assembly. It should be noted that the conical and cylindrical modes of vibration are nearly coincident. The predicted stability characteristics of the proposed Phase II rotor assembly is shown in Figure 9. Comparison of Figures 5 and 9 indicates that the stiffness required for stability was reduced from 2,000,000 lb/in. to 350,000 lb/in. at the design speed.

A comparison of important journal bearing characteristics for the Phase I and Phase II bearings is presented in Table II. This table shows that the proposed bearing will have a better chance of meeting the design requirements for these reasons:

35% lower rotational speed than the original design.

7% lower surface velocity than that already demonstrated during Phase I at 71,500 rpm.

56% lower stiffness than that demonstrated during Phase I.

33% reduction in supply pressure.

33% reduction in shaft diameter compared to the original design.

TABLE II

COMPARISON OF PHASE I AND PHASE II BEARING DESIGNS

	PHASE I BEARING		PHASE II
	DESIGN	TESTED	BEARING
Maximum Speed, rpm	135,000	71,400	100,000
Diameter, inches	1.875	1.875	1.25
Temperature, °F	1000	200*	1,000
Radial Clearance, inches	0.0004	0.0009	0.0005
Surface Velocity, in/sec.	13,200	7,030	6,550
Stiffness, lb/in.	2,000,000	800,000	350,000
Supply pressure, psia	1,500	1,200	800

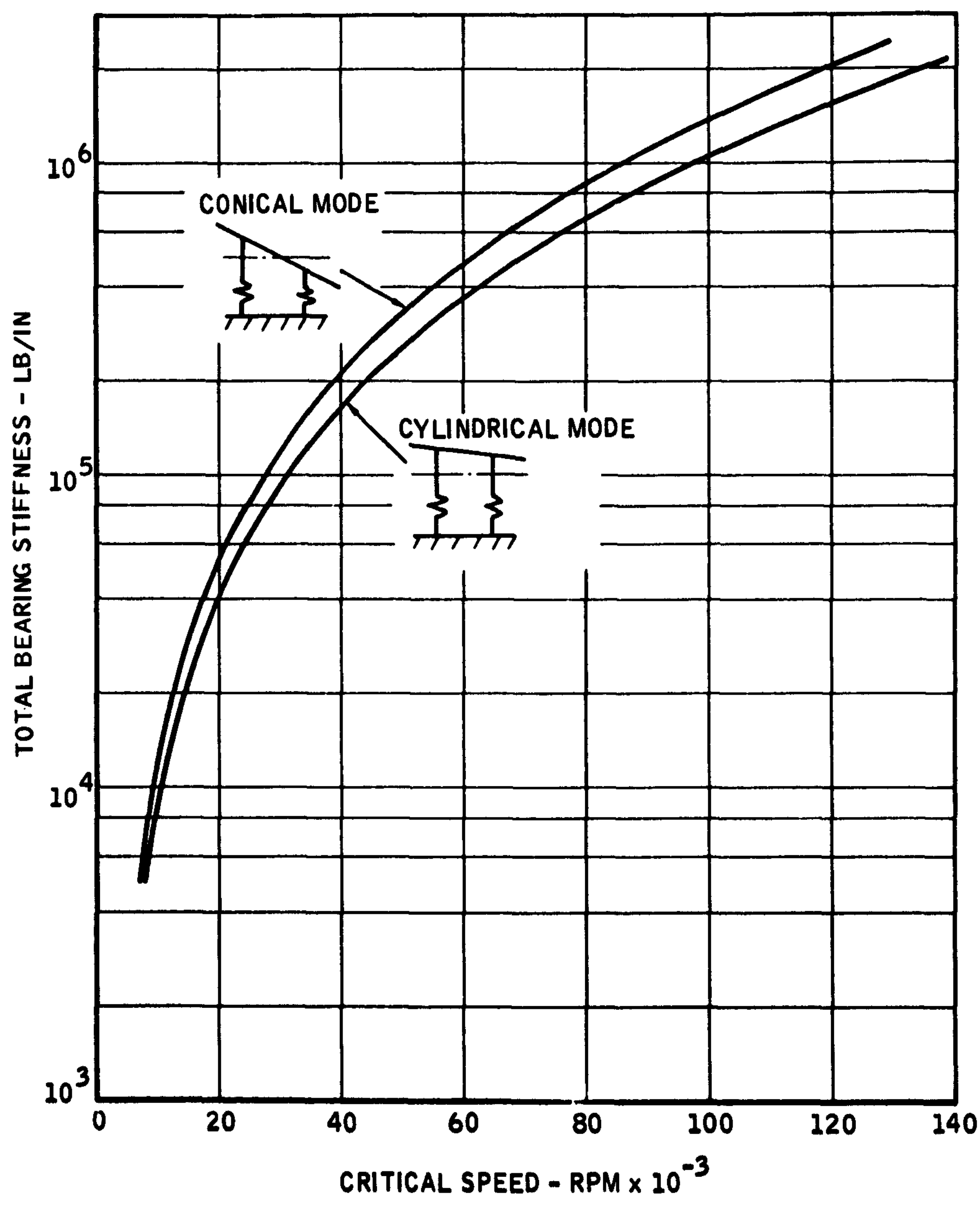

Figure 8. Phase II Air Bearing — Bearing Stiffness vs. Critical Speed

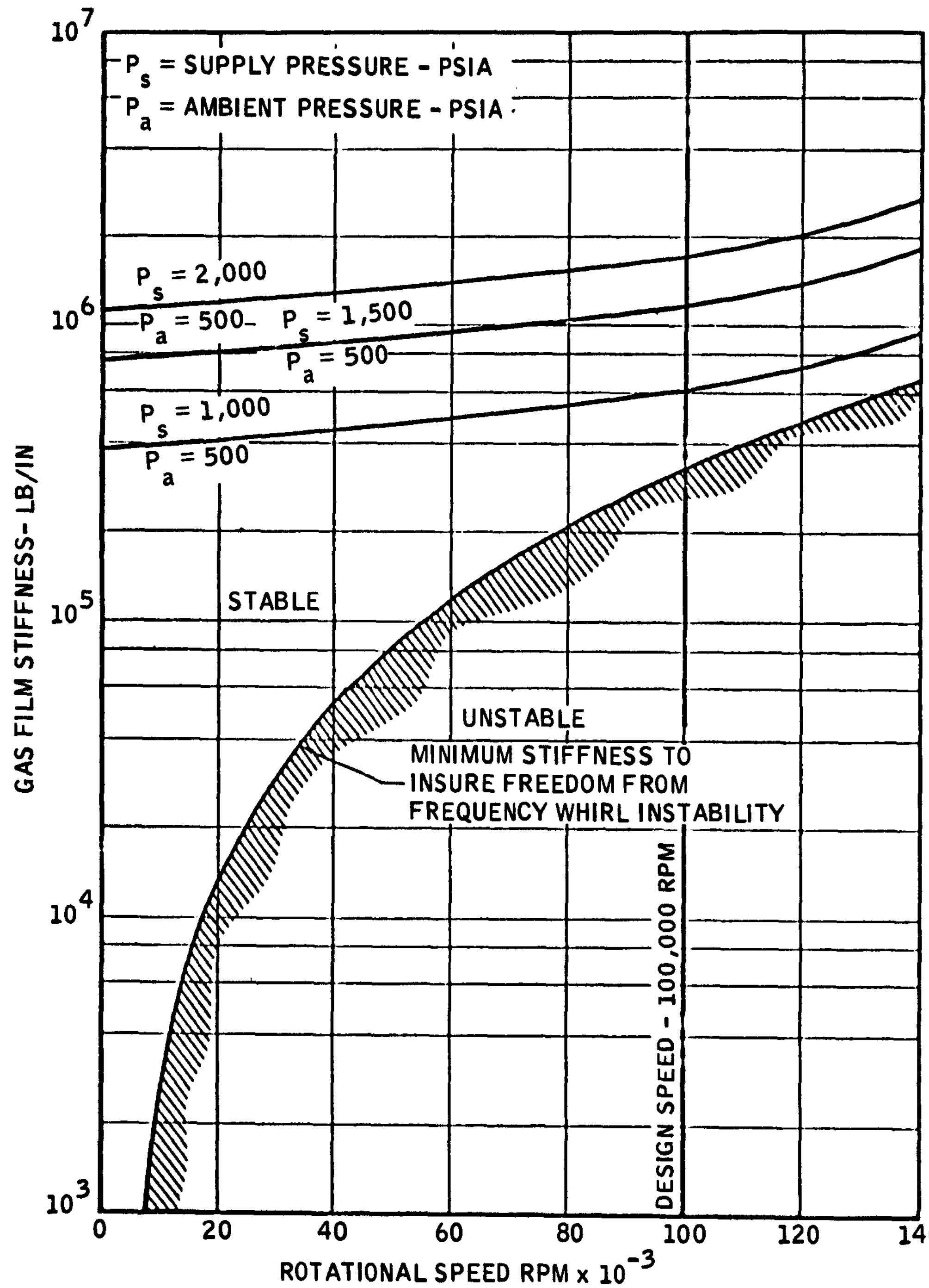

Figure 9. Predicted Stability Characteristics of Phase II Air Bearing

Conclusions

The intent of the Compressor Experimental Development Program, Phase I, was satisfactorily completed during the period covered by this report. The air bearing configuration is defined. Interstage labyrinth sealing is within the present state of the art: no preliminary development is required. The compressor design performance was satisfactorily demonstrated. It is, therefore, recommended that Phase II be initiated to evaluate the experimental ram air turbine driven compressor in the extreme thermal environment.

ROTARY ACTUATOR AND SERVO VALVE

Fabrication and assembly of the component parts for one experimental actuator and servo valve were completed. This includes the following sub-assemblies shown in Figures 10, 11, 12 and 13: rotary actuator, pilot amplifier (first stage), plate valves (second stage), and feedback valve.

The feedback valve consists of two orifices in series; the upstream orifice is fixed and the downstream orifice is variable. The downstream orifice, which exhausts to atmosphere, is controlled by a cam attached to the output shaft of the rotary actuator. The position of the cam determines the control pressure generated by the feedback valve. Figure 14 is a photograph of the feedback valve, test fixture, and calibration dial used during initial calibration tests.

Preliminary experimental testing of the rotary actuator and servo valve is now underway. This will include the following tests at room temperature and full load: stability, input signal vs. position, and transient response.

PRESSURE REGULATOR AND RELIEF VALVE

The pneumatic pressure regulator design was modified due to problems reported previously in reference 1. Testing of the main stage was satisfactory up to 1300°F. A failure of the main bellows assembly precluded further evaluation

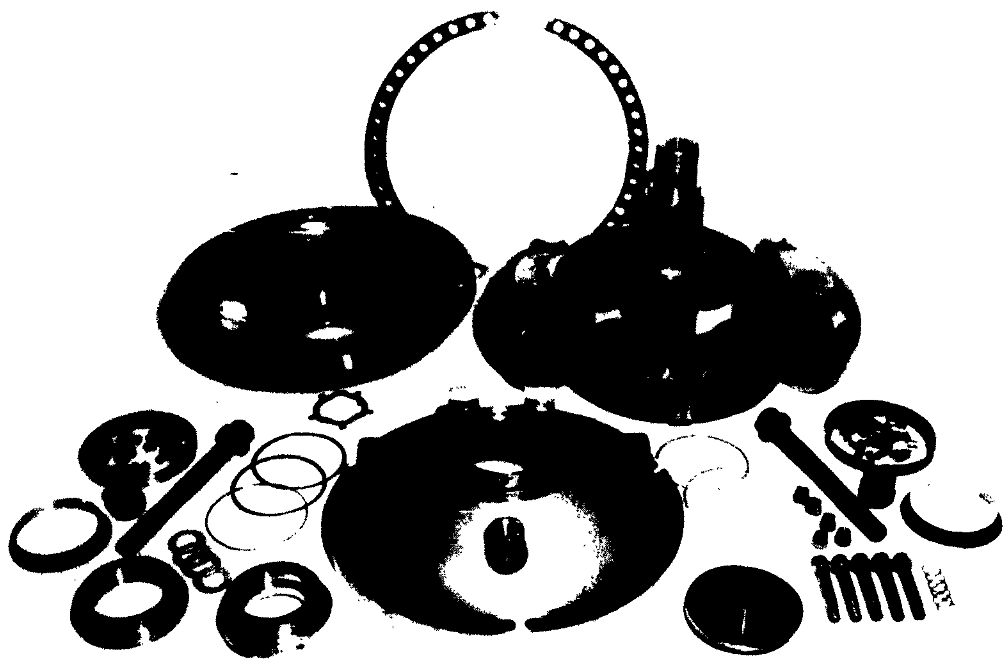

Figure 10. Rotary Actuator Assembly

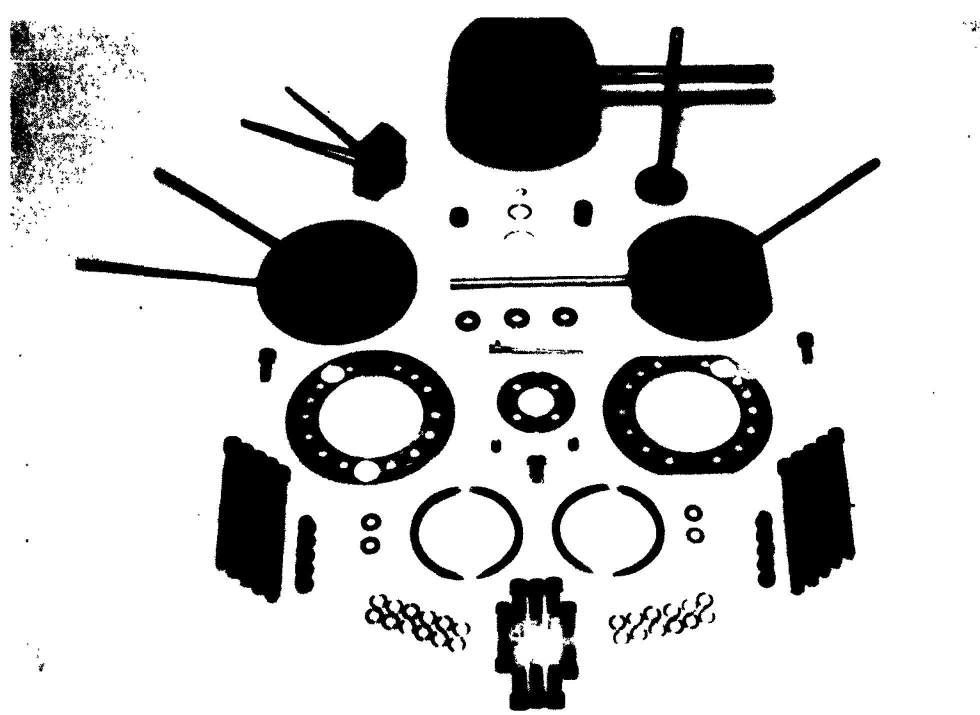

Figure 11. Pilot Amplifier (First Stage) Subassemblies

Figure 12. Plate Valves (Second Stage) Subassemblies

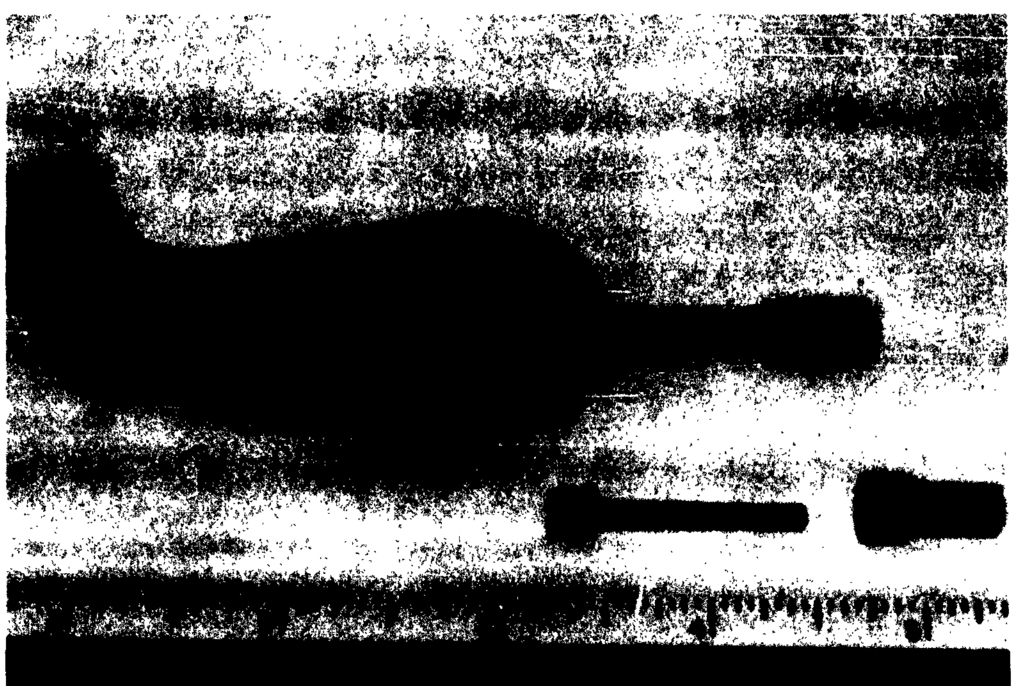

Figure 13. Position Feedback Valve

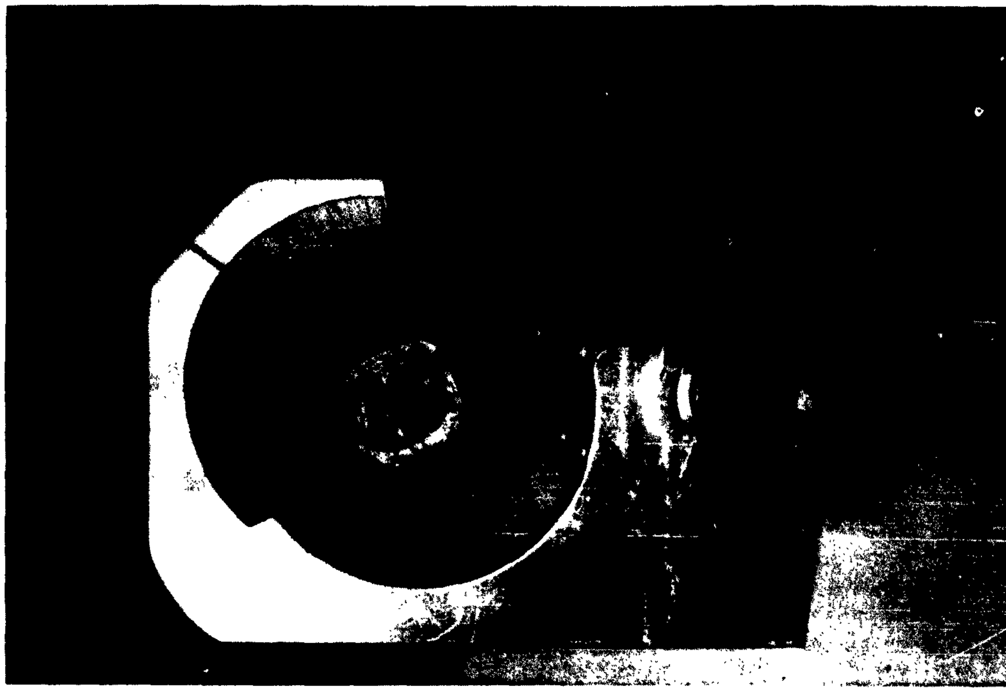

Figure 14. Position Feedback Valve Calibration Test

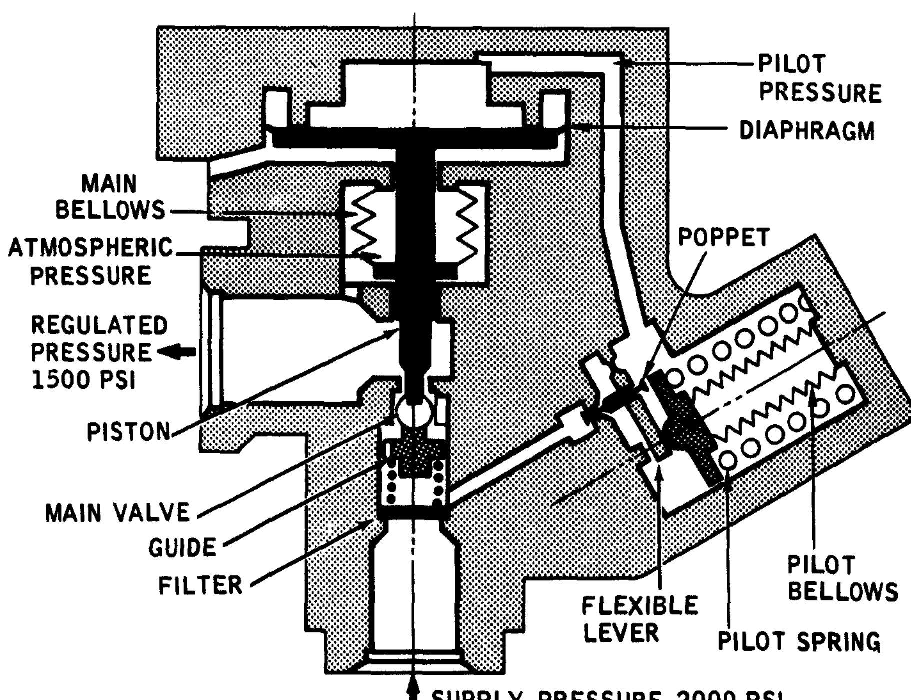

Figure 15. Pneumatic Pressure Regulator (1500° F)

at higher temperatures. In regard to the relief valve, two Model F units are being fabricated with delivery scheduled for 30 May 1963.

Pressure Regulator

The revised pressure regulator design is shown in Figure 15. The unit shown incorporates the changes to the main and pilot stages discussed in the last Quarterly Report (ref. 1). Relaxation of the main stage piston rings required the substitution of a René 41 diaphragm in place of the piston. The pilot stage was also modified to incorporate a René 41 spring in conjunction with a gas-filled bellows. A flexible lever arrangement was also added in order to reduce sliding friction (resulting from side loading) and increase the flow capability of the pilot stage.

The main stage diaphragm assembly was fabricated and experimental testing initiated. During the first test, the assembly performed as expected up to a temperature of 1200°F. At this point, however, the assembly stopped modulating and remained in the open position. The ambient temperature was reduced to room temperature and the unit was disassembled. The piston rod (see Figure 15) had seized, and was frozen in its guide; excessive galling was noted between these René 41 components.

To eliminate this problem the guide was revised to incorporate a spherical radius (this results in line contact in place of cylindrical contact). Also, a Lubco coating was applied to both mating surfaces in order to reduce galling. Testing of this revised main stage configuration was then resumed.

Performance of the unit was satisfactory up to 1300°F when a second failure occurred. The unit was returned to room temperature and disassembled. The René 41 bellows had ruptured. The steps taken to prevent galling between the guide and piston proved to be successful; there was no evidence of excessive friction or seizure.

A new bellows is being manufactured and parts for the pilot regulator are nearing completion. Testing of both the main and pilot regulator stages will start once these parts are completed.

Relief Valve

Model X relief valve testing was previously completed. The design of the Model F unit was completed during the period covered by this report. The basic change involves the incorporation of a "kidney" seal design in order to reduce the external leakage. The parts for two Model F relief valves are being manufactured. Delivery is scheduled 30 May 1963.

ACCUMULATOR

The accumulator forgings were given an initial heat treatment and were then machined. Three accumulators were electron beam welded to form pressure vessels. The three units were X-ray inspected subsequent to heat treatment; no flaws were present in either the welds or the heat affected zones. The following heat treatment and sequence of operations were performed to minimize grain growth:

1. Anneal forgings at 1550° F/4 hours, air cool.
2. Machine forgings.
3. Electron beam weld.
4. Solution heat treat at 1975° F/4 hours, air cool.
5. Double age.
   (a) 1550° F/24 hours, air cool.
   (b) 1400° F/16 hours, air cool.

This is a modification of the original sequence outlined previously (ref. 1). Welding of two Model X (experimental) units in the solution-heat-treated condition resulted in a relatively coarse grained structure. A finer grained structure is expected to improve ductility and fatigue strength.

Some difficulty was experienced in locating a facility for heat treating the units in a hydrogen atmosphere. This resulted in a delay of approximately two weeks. Delivery of the first completed assembly is now scheduled 29 March 1963.

FILTER

The two pneumatic filters were repaired by Bendix Filter Division and shipped to Convair for evaluation. Excessive external leakage was encountered on S/N 1 subsequent to operation at 1000°F.

Both units were subjected to a proof pressure (3300 psi) and a leakage test (2000 psi) at room temperature. The results were satisfactory in that no external leakage was observed. This was followed by high temperature testing.

The first unit was installed in the environmental chamber and the temperature was increased to 1000°F. A pressure of 1000 psig was applied in order to check out the system. External leakage was noted. The unit was cooled to room temperature and a leakage check was performed at 2000 psi inlet pressure. This was done by measuring the pressure drop of a fixed volume over a two minute time interval.

The ambient temperature was then raised to 1500°F and held at this temperature for one hour prior to performing a second leakage check. A proof pressure test was also performed while at this temperature. The unit was then cooled to room temperature and a third leakage test was performed. The second unit was tested in a similar manner except that the leakage test at 1500°F was performed at 3300 psig inlet pressure. The results of this testing are summarized in Table III.

TABLE III

PNEUMATIC FILTER EXTERNAL LEAKAGE TEST RESULTS

External Leakage, lb/min.

Unit 1			Unit 2		
Room Temp. (Post 1000°F)	1500°F	Room Temp. (Post 1500°F)	Room Temp. (Initial)	1500°F	Room Temp. (Post 1500°F)
0.14	0.10	0.16	0	0.0076	0.0012

Specification requirement: 0.0026 lb/min. (max) at 2000 psig.

It should be noted that the leakage recorded in Table III includes that due to both tube fittings and boss seals incorporated into the test setup. Although every effort was made to eliminate fitting leakage at room temperature, the actual source of leakage at 1500°F, could not be established. The source of leakage was located both before and after operation at high temperature. Since the method used in determining the leakage is approximate, a more accurate method was used to check the values obtained at room temperature. Each unit was installed in a sealed inclosure and the external leakage determined using a flowmeter. The data obtained was of the same order of magnitude as that obtained using the pressure drop method.

Examination of Unit No. 1, after being subjected to 1000°F, revealed that the external leakage was coming from the welded zone in the filter head assembly. This is the same problem encountered previously (ref. 1). It is anticipated that this unit will be returned to Bendix Filter Division for rework. In the case of Unit No. 2, the leakage was located as originating from both the welded zone and the seal between the bowl and head. In this case, however, the leakage is acceptable.

4 | MAJOR PROBLEM AREAS

During the initial stages of this development program it was recognized that a number of problem areas existed. Resolution of these would, to a great extent, effect satisfactory completion of the program goal: development of a pneumatic system capable of sustained operation in extreme thermal environments. In general, the following were recognized as major problem areas: springs and bellows, bearings, seals (dynamic and static), material wear properties, and the joining of superalloys. These problems, as well as their solutions, are discussed in this section.

The design of springs and bellows for use at high temperature must be governed by both stress relaxation effects and change of the elastic modulus of the material as a function of temperature. Although springs are still considered a potential problem area, some success was achieved in obtaining satisfactory springs for both the check valve and the relief valve. This was accomplished by using a heat setting process and operating at a reduced stress lelve. The maximum anticipated relaxation is 25% after a 70-hour exposure of 1500°F.

Fabrication difficulties were encountered during construction of the compensator bellows used in the pneumatic pressure regulator. The basic problem, concerned with obtaining satisfactory welds of the ultra-thin (0.003-inch) gage René 41 material, was not resolved. Two bellows, of 0.006-inch gage René 41, were fabricated. These have been subjected to repeated thermal cycling with good results. Although leakage was subsequently experienced, the bellows were repaired and again functioned satisfactorily.

The development of suitable bearings for extended operation at 1500°F continues as a possible problem area. Spherical roller bearings were selected for use in the rotary actuator. Both races and rolling elements were fabricated from René material while the contact surfaces were coated with ceramic-bonded calcium fluoride. Performance of these bearings, evaluated in conjunction with wear testing conducted on the rotary actuator, was judged satisfactory. A gas film lubricated bearing was selected for the compressor. Testing to date has been accomplished at speeds up to 72,000 rpm with satisfactory results.

Seal design is instrumental in determining overall system efficiency. During the course of this program a satisfactory metal boss seal was evaluated. The piston ring type dynamic seal was evaluated during testing conducted on the rotary actuator and the pressure regulator. A total of 888 cycles, over a period of 15 hours at 1500°F, was completed during wear testing of the rotary actuator. In the case of the pressure regulator, a similar type of dynamic seal was unsatisfactory due to the more stringent leakage requirements for this component.

Some inroads were made in resolving the problem associated with material wear properties. The relief valve incorporates a solid chromium-carbide poppet vs. a Haynes alloy No. 25 seat. This combination was successfully evaluated from room temperature to 1500°F. A ceramic-bonded, calcium-fluoride coating was evaluated during wear testing conducted on the rotary actuator with good results.

Joining of the superalloys is also considered a major problem area due to difficulties encountered during fabrication of the accumulator and filter elements. A satisfactory weld of the Udiment 500 accumulator was obtained using the electron beam process. In the case of the filter element, all attempts to vacuum sinter the René 41 filter element were unsuccessful. Good results were subsequently obtained using Hastelloy X wire in combination with a René 41 core. The accumulator is scheduled for evaluation during the next quarterly period.

5 | FUTURE WORK

Testing and evaluation of components, for subsequent inclusion in the high temperature pneumatic system, will continue. In this regard, experimental testing of the pneumatic filters is now in the final stages. Evaluation of the rotary actuator and servo control valve package is also scheduled for the forthcoming quarterly period.

This experimental testing is required in order to determine the limitations of pneumatic type fluid power control systems. Recent advances in materials and techniques are incorporated into these components. It is, therefore, necessary to evaluate each component prior to subjecting the entire pneumatic system to the high temperature environment.

REFERENCES

1. "Development of a High-Temperature, Nuclear-Radiation-Resistant Pneumatic Power System for Flight Vehicles," General Dynamics/Convair, GD/C 63-001, (24 December 1962).

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